Improvements in Reciprocating Compressors - With Special Emphasis on Interesting Developments for Refrigeration in the G.D.R

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ABSTRACT

The serie of G.D.R. hermetically sealed, semi-hermetically sealed and open-type refrigerant compressors are introduced. In using the complex simulation to increase the coefficient of performance the further development of hermetically sealed refrigerant compressors is discussed. Considering the real conditions of application during intermittent operation in the refrigerators the compressors are optimized by dynamically determining the temperature field. The optimization of the stroke/bore ratio ist described by a computational model for semi-hermetically sealed and opentype refrigerant compressors, and special constructive solutions to decrease the heating of the suction gas, to guarantee a sufficient internal sealing, to lower frictional losses, to secure the oil supply and an automatic stop valve are dealt with.

SURVEY OF SERIES OF REFRIGERANT RECIPROCATING COMPRESSORS IN G.D.R.

In the GDR, refrigerant reciprocating compressors were mainly produced in three owned enterprises (VEB). As it is shown in figure 1, the following three series are produced:

- hermetically sealed compressors (h) in eight sizes, rate of delivery: 0,52 - 2,5 m^3/h;
- semi-hermetically sealed compressors (sh) in eight sizes, rate of delivery: 10 - 112 m^3/h, among them the four large sizes are also produced as open-types and two sizes as twostage types;
- open-type compressors (o) in eight sizes, rate of delivery: 160 - 1600 m^3/h, among them the six small sizes are also manufactured as semi-hermetical types and two sizes as twostage types.

In these enteprises, refrigerant reciprocating compressors are traditionally produced so in oneof these enterprises compressors have been prdouced for more than 120 years. In the last ten years, the series above mentioned were constructed based on this experience, and the theoretical volume flow rates during suction were classified on the basis of the standard series R 5 / R 20 agreed upon in the RGW (Council for Mutual Economic Assistance), and the strong demands

- to lower the power consumption

- to decrease the materials' utilization
- to increase reliability and to reduce repair and maintenance

were largely met. These series of compressors were designed and constructed according to their special use.

The hermetically sealed compressors were developed for LBP and MBP application, the refrigerant R 12 to be used in domestic refrigerators, commercial refrigerators and in freezing cabinets. For HBP application including heat pump appropriate modifications are at disposal. In normal climate (N) the hermetically sealed compressors are usable, in tropical climate appropriate models are offered which are operated by different voltages and frequencies. At present 1,5 million hermetically compressors were produced in the G.D.R. annually.

The semi-hermetically sealed compressors of the series 50/35 - 2 and 60 - 4/2 are suitable to be used with the refrigerants R 12, R 22, R 502, R 13, R 1331 and R 114. They are applied in cascade or two-stage low temperature refrigerating installations, in normal climate refrigerating plants, in air conditioning plants and in heat pump installations. Because of its special use for truck refrigeration systems the enclosure of the crank and motor of the series 60 - 4/2 were made of light metal.

Then open-type compressors of the series BR 5 are especially intended for the refrigerant ammonia and for cold stores, slaughter houses, processing plants of the foodstuffs industry, in chemical enterprises and in sports' buildings. The semi-hermetically sealed compressors of this series are outlined for the refrigerant R 22 and R 12 and are especially used for chilled waters, heat pumps and refrigerating installations in ships.

Accordingly, the GDR disposes of an extensive range of refrigerant reciprocating compressors, meeting virtually all the demands of the different fields of application. In the following, some of the lately important improvements and further developments in the field of these compressor series are exemplified.

APPLICATION OF THE COMPLEX SIMULATION OF HERMETICALLY SEALED COMPRESSORS WITHIN THE FRAME OF DEVELOPMENT OF COMPRESSORS

Since the seventies, the instruments of development are completed by the simulation technology. When further developing hermetically sealed compressors, in addition to extensive experimental investigations, there were applied computational programs for the complex static simulation of hermetically sealed compressors, see figure 2. In the program to determine capacity, the process in the cylinder of the compressor including the pressure losses in the lines and dampers, the mechanical lost power and the lost power of the motor are calculated. The thermal simulation comprises the computational modeling of heat transfer processes. For the optimization a network of the internal power transport (NET-model) consisting of 34 elements is used. The simulation program ist outlined for digital computers in FORTRAN language.

In the optimization we proceeded from the following targets:

- to attain a maximum C.O.P. in LBP application, a high volumetric efficiency, especially at low evaporation temperatures is reqzired.
- to attain a maximum C.O.P., minimal indicated mechanical and electrical

losses are required.

To attain a volumetric efficiency as high as possible, the vapour of the refrigerant should be passed into the cylinder as cold as possible, the back streaming losses should be as low as possible, the valves should be tight during the closing and the clearance volume should be as small as possible.

Consequently, the following conclusions resulted for the outline, which were obtained by the construction:

- The suction gas was not more used to cool the motor, this is done by the oil.
- The stroke/bore ratio was optimized considering the main field of application.
- The area of the cylinder head / valve plate was designed to secure a small clearance volume, tight seal and to avoid heat transfer to the suction.
- The suction gas was led in such a way that only a low heating in the capsule occured.

To decrease the mechanical losses, the design of the bearing was of great importance. Extensive investigations werte carried out concerning the outline of the crank assembly, the geometry of the bearings, the surface finish, the bearing clearance and the oil supply. Within these investigations oil of different viscosity was considered.

The improvements obtained by decreasing different losses are illustrated in figure 3. In the small sizes, with a rate of delivery of 0,5 - 0,8 m^3/h (piston displacment 3 - 4,5 cm^3) the C.O.P.is increased by lowering the mechanical, indicated and motor losses by about 6 % respectively, though in this low rate of delivery to decrease the heating of the suction gas was difficult. As to the large sizes, the C.O.P. was improved by about 30 %. This resulted in decreasing the mechanical losses by about 13 % and by lowering the indicated losses by about 17 %. To lower the heating of the suction gas was of great importance for these compressors. In figure 4 it is shown how the temperature level of the larger compressors was influenced by those energetic improvements. Obviously, the temperature lowering in front of the suction valve as a result of the demand to improve the volumetric efficiency also resulted in a temperature decrease in the compressor, positively influencing life and reliability. Compared with the temperature at the cylinder outlet, the temperature at the pressure connection should only decrease to a small extent in order to transport by the pressurized gas as much heat as possible out of the case. It was evident, how under standard regime the energy losses of a hermetically sealed compressor are variable.

When actually applying the compressor in the refrigerating plant no stationary regime is valid due to intermittent operation, a dynamic simulation of the compressor is required. An important aspect of such a model is the dynamic calculation of the temperature field in the compressor differing from the stationary in that the heat capacity is considered in the individual elements of the NET - model. The program system for the dynamic simulation offers the possibility to optimumly adjust the compressor to the refrigerating plant in varying the differenz optimizing parameters. In figure 5 the results of the dynamic simulation are compared with the measured values.

SPECIAL CONSTRUCTIVE SOLUTIONS IN THE 50 / 35-2 AND 60 - 4/2 SERIES OF SEMI-HERMETICALLY SEALED COMPRESSORS

In figure 6 the construction of semi-hermetically sealed compressors of the 50 / 35-2 series is illustrated. Four sizes were produced with two strokes of 35 and 50 mm and with three cylinder diameters of 45, 55 and 65 mm as a two-stage type.

To cool the compressors suction gas cooling, static and dynamic ventilation of the crankcase/cylinder head area as well as water cooling of both covers of the casing are provided, depending on the conditions of application. Preferably, the suction gas cooling is used for evaporation temperatures up to -20 ^{o}C. In figure 7 the limits to be considered in applying the different types of cooling are represented.

The reed valves are mounted on a joint valve plate for both cylinders. A maximum flow area was obtained to minimize the pressure losses by the U-shaped suction valve-reed in the pocket of which there are two bores of delivery valves covered by two individual reed valves, see figure 8. On the basis of extensive experimental investigations of the valve lifting behaviour and of the capacity of the compressor the valve lift and the reed thickness were adjusted in dependance of the operational regime, so that the pressure loss, sound level and load are an optimum. Regarding a minimum impact load and flexural load special value was set on the constructive outline of the place to chuck the reed and of the stops of the suction valve reed.

By testing several variants within extensive series of experiments special was paid to the secure oil supply. For all compressors cooled by suction gas and for compressors cooled by suction gas and for compressors with external cooling an optimumly designed disk to transport oil was chosen as an uniform version in contrast to a non-uniform version made by other manufacturers, see figure 6. The oil swept by the disk and centrifuged into the internal shape of the cover is collected in the pocket of the cover, enters the concentric bore of the eccentric shaft and supplies the steady bearings and connecting-rod bearing. When the oil circuit of the refrigerant plant is mastered in all operational phases, there is sufficient oil present. If it is not the case from the beginning, a type with oil pump is offered and the oil supply is controlled by an oil pressure difference pressostate. The non-return valves found to some extent with other manufacturers between the crankcase and the motor casing, the equipment for oil transport from the motor casing to the crankcase as well as on mechanisms for oil charge into the eccentric shaft are superfluos, consequently, a very simple system of oil supply was chosen.

The 60-4/2 series comprises four sizes of semi-hermetically sealed and open-type construction with one stroke and two cylinder diameters of 70 and 80 mm in a two-cylinders and four-cylinders' outline, see figure 9.

The valves are designed as concentric ring plate valves. The suction gas flow was outlined under the aspect of the smallest possible joint diaphragm of suction and discharge to keep the heating of the suction gas as low as possible. The difficult tolerance compensation between the internal and external series of tolerances was obtained by a ring plate spring (Belleville spring) of a patented type. In addition to the favourable thermal behaviour this valve layout is advantageous expecially because of the more simple mode of operation during repair and maintenance and in reliably sealing the cylinder head to the outside only by one seal between the casing and the cylinder head.

As it is conventional in this range of capacity, an oil pump provides for the oil supply, which lies low in the oil sump. The direction of delivery is maintained by reversing valves when the sense of rotation of the motor is changing. To shorten the time required to establish a stable oil pressure after putting the compressor into operation, especially when accumulating the refrigerant in the oil, on the extension of a gear-pinion shaft of the oil pump a little propeller (vortex) was mounted. By this patented solution the oil is moved and the solved refrigerant escapes very fast. Thus, when starting the compressor, wear was considerably reduced and life of the driving mechanisms was prolonged.

The four cylinder compressors of this series to a great extent are produced in a two - stage version, whereby three cylinders are operating in the first stage. This type can be used in many fields of application to refrigerate foodstuffs, otherwise requiring two separated compressors. Because of the favourable heat dissipation of this cylinder head by air cooling or water cooling an intercooler is not required. In addition to the enlargement of the limits of application the refrigerant flow sucked from the first stage because of the low pressure ratio is improved, and in spite of the lower theoretical volume flow rate (piston displacement) the cooling effect increases as compared to the one-stage type. In addition, the energetic efficiency (C.O.P.) is considerably increased by the two stages, see figure 10. In the two-stage type the small end bearing of the second stage is especially stressed, because it is not revealed by the intermediate pressure at charging stroke, consequently very few oil reaches the stressed area of the bearing. To avoid this situation, this bearing is designed as a needle bearing without cage, and now at any time sufficient oil can enter there.

To reduce the harmful influence of the refrigerant solved in oil during stop of the compressor, in general three measures are taken:

- during stop the suction valves and discharge line valves of the compressor are closed,
- the oil heating is switched on during stop
- or it is required to switch on for exhausting.

In case of transport refrigeration (road vehicles, railway), in general none of the three safety variants is realizable. For those applications the patented automatic shutoff valve was developed, see figure 11. This valve is to be used together with a compressor and an oil pump, as under regime without any oil pressure both pistons separating the suction and delivery of the compressor from the refrigerating plant are pressed by elasticity onto their bearing area, and thus perfect sealing is obtained by the PTFE rings inserted into the pistons. After the start the compressor is running in a balanced form over the bypass built in in the automatic shutoff valve until the oil pressure was set up and the sucking and discharging pistons are opened by the hydraulic piston and the servo-contrelled valve and the bypass is closed. Thus, the reliability of the compressor is considerably improved.

OPTIMIZATION OF THE STROKE/BORE RATIO AND SPECIAL CONSTRUCTIVE SOLUTIONS CONCERNING SEMI-HERMETICALLY SEALED AND OPEN-TYPE COMPRESSORS OF THE BR 5 SERIES

To optimize the stroke/bore ratio when the BR 5 series was further developed, a computing program to simulate the working cycle of a reciprocating compressor was elaborated, In considering definite presumptions, we proceeded from the following equations:

Equation of the pressure behaviour in the cylinder

$$dp = np\left[\frac{dm}{m} - \frac{dV}{V}\right] \qquad (1)$$

Equation of the volume's alteration

$$dV = A_K \cdot r\left[\sin\alpha - \frac{\lambda s}{2}\sin 2\alpha\right] d\alpha \qquad (2)$$

Equation of the flow in the valves

$$dm = \varepsilon \cdot \alpha_d \cdot A_d \sqrt{2 \cdot \varrho (p_1 - p_2)}\, d\tau \qquad (3)$$

Equation of the valve's movement

$$\frac{d^2h}{d\tau^2} + \frac{dh}{d\tau} k_1 \cdot 2 \cdot \xi + k_1^2 \cdot h = k_2 \qquad (4)$$

Concerning the pressure difference under the radical of equation (3), we distinguish between four cases

1. suction $p_1 - p_2 = p_S - p$
2. discharge $p_1 - p_2 = p - p_D$
3. return flow (suction valve) $p_1 - p_2 = p - p_S$
4. return flow (discharge valve) $p_1 - p_2 = p_D - p$

To calculate the valve's movement, in the equation (4) the following forces were taken into consideration:

- force due to the pressure difference existing on the valve
- inertial force of the valve mass
- dead weight of the spring-mass-system
- elasticity
- damping force
- retentive force at the beginning of the opening process.

The rebound vibrations were comprised by the coefficient of recovery ψ.

At first, the differential equations were transformed into an integrated form, this was easy to realize in the non-substituted form of the equation. As in solving the differential equations the coefficients k_1 and k_2 are considered to be constant, the validity of equation (4) can only be solved successively. This is also valid in calculating the cylinder mass resulting from addition of individual subsets. The result of calculating by means of the digital mini computer C 8205 was a table including the crank angles, piston displacement, the pressure of the cylinder, the valve lift and valve rate. To determine the capacity, the area of the p,V-diagram was calculated.

Due to the structure of the mathematical model a realistic simulation of the pressure behaviour and of the valve's movement is only possible when certain system's parameters are introduced.

The flow pressure coefficient ϱ_{Str}, the coefficient of recovery ψ, and the damping coefficient ξ decisively influence the behaviour of the valve lift curve. Each of these parameters characteristically influences the valve lift curve, see figure 12a, b, c. To optimize the behaviour of the valve lift curve it is important to know the influence of certain parameters such as the spring constantc, the pretension of the valve spring h_o, and the mass of the valve plate, this can be achieved by means of the simulation - computing program, see figure 12d, e, f. To compare them with p,V-diagrams and valve lift diagrams and with experimentally calculated diagrams measurements were made at a test stand. In figure 13 a good conformity of the calculated results and experimental results is shown.

On this basis, the constructing engineer is enabled to design a valve ensuring an optimum curve of the valve lift. The method of calculation presented comprises many simplifications and restrictions. To investigate the effect of these simplifications an extensive and complicated computing program was set up considering the influence of the vibrations in the suction and discharge lines, the heat transfers in the cylinder, the change of state of the real gas as well as leakage losses by the piston rings and closed valves. Calculations for comparison on the BESEM 6 have shown that the valve lift behaviour is only more influenced by the gas vibrations, other influences have a low importance.

The interweaving of the factors influencing the stroke/bore ratio are of a great complexity. In order to give an aid to the constructing engineer to roughly select the stroke/bore ratio and to quantitatively estimate the results of alterations in this ratio and their influence on the C.O.P. and on other parameters, a nomogram of the following most important influencing factors was set up: restrictor losses in the valves, compressor speed, heat to be dissipated from the cylinder. In varying the stroke and on the basis of statistics a curve for the optimum of C.O.P. based on calculations with the simulation - computing program was plotted in the nomogram.

The compressors of the BR 5 series were designed on this basis for average conditions: evaporation temperature: -15 °C, condensation temperature: +35 °C, refrigerant R 22 for an universal application in the low temperature range, normal temperature range and in the conditioned range, their stroke/bore ratio was 0,8 for 100mm Ø and 0,75 for 200mm Ø.

The optimization carried out by simulating the working process was to be completed by determining the mechanical lost power. The lost power due to friction was calculated on the basis of the p,V-diagrams, whereby average constant pressures were used for the suction and discharge phases. The overall lost power due to frictions is composed of the following elements:

- friction at the piston by lateral forces, due to gas- and mass forces,
- friction at the piston ring, due to gas forces and self-contained stress of the piston rings,
- friction at the radial bearing, due to gas- and mass forces,
- friction at the axial bearing, rotary seal,
- power consumption of the oil pump,
- losses, due to ventilation.

As a calculation's result on the BESEM 6 the piston rings cause the greatest deal of the friction. Second comes the loss due to friction in the rotary seal and in the sleeve bearings. On this basis the mechanical losses were optimized on the 8-cylinder compressor, the diameter of which being 200 mm, consequently, the friction was decreased from 15,7 kW originally to 12,7 kW. Thus, 3 kW were economized and this corresponds to a lowering in friction of 19 %, see figure 15.

In addition, in developing the BR 5 series it was intended to improve the internal seal by taking constructive measures, to decrease on this way the heating of the suction gas before it enters the cylinder and to obtain a better volumetric efficiency and a reduction in thermal load of the compressor. In the BR 2 series of compressors produced before the seal between the cylinder and the suction chamber was obtained by two metallic sealing strips mounted on one level, between those strips are the bores for the passage of the suction gas. For this layout there arose problems as to the grinding in of those metallic sealing surfaces and a poor seal was the result, see figure 15b. In the BR 5 series only one sealing strip was attached between the cylinder and the suction chamber to seal and only one round rubber gasket to seal between compression and suction chamber, see figure 15a. It was shown by comparative experiments, that the suction gas temperature in the BR 2 series increased by about 70 K when entering the suction valve until arriving at the suction valve plate, however, as to the BR 5 series the temperature increased only by 10 K.

Compared with the BR 2 series the C.O.P. of the BR 5 series was considerably improved by applying the measures pointed out. At a evaporation temperature of -15 °C and at a condensation temperature of +35 °C the C.O.P. of compressors of 100 mm diameter cylinders is increased by about 15 % and for compressors of 200 mm diameter cylinder it is increased by about 20 %.

SUMMARY

In newly and furhter developed series of refrigerant reciprocating compressors in G.D.R. the power consumption was considerably reduced. As it follows from the measures explained this was achieved by the application of the simulation computer technique, by experimental investigations and by satisfactory constructive solutions. The instruments to develop reciprocating compressors are completed by the simulation computer technique and thus the possibility was brought about to optimize compressors, however, it can not be done without experimental investigations, and the constructing engineer with his wealth of ideas will play the principal, if not the decisive part.

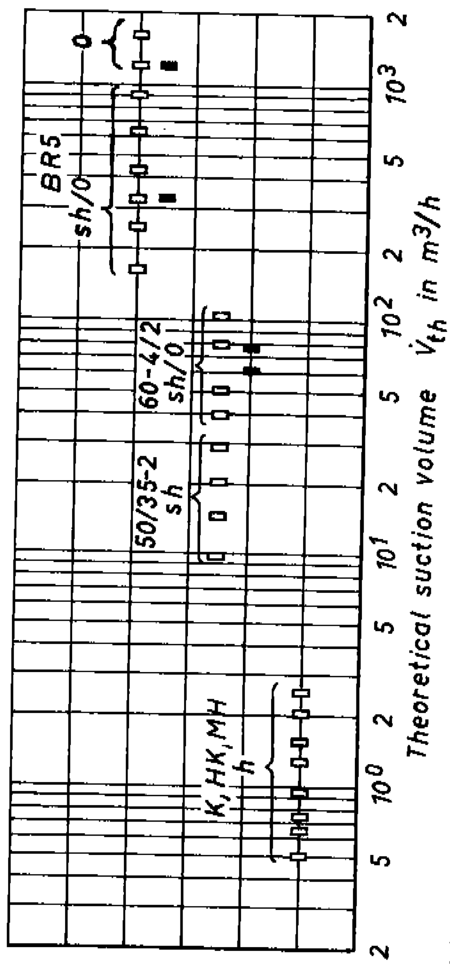

Fig.1: Series of reciprocating compressors for refrigeration in the G.D.R.
h hermetically, sh semi-hermetically, o open,
□ single-stage, ■ two-stage

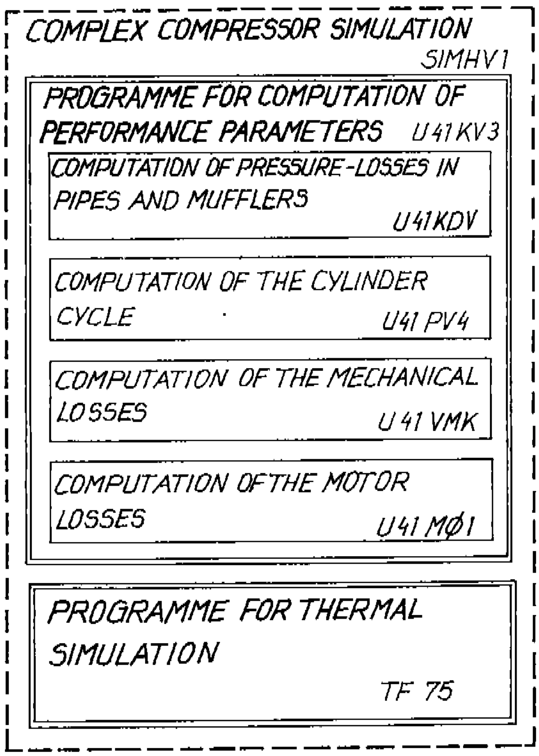

Fig.2: Structure of the simulation programme

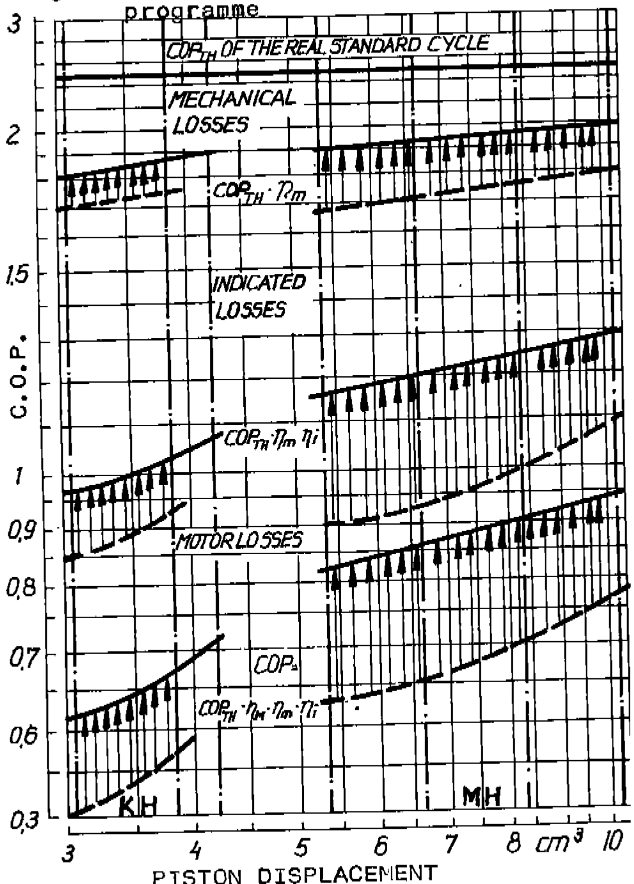

Fig.3: Improvement of the C.O.P. for hermetically compressors types KH and MH

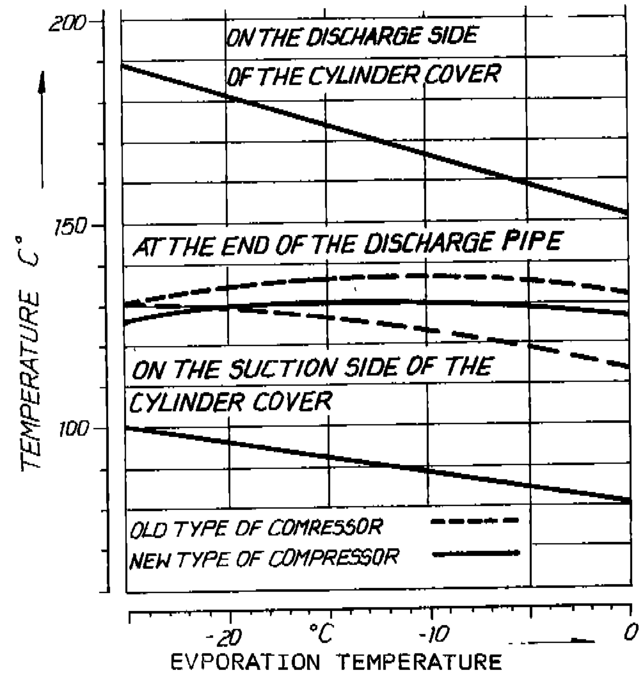

Fig.4: Characteristic temperatures of different hermetically compressors

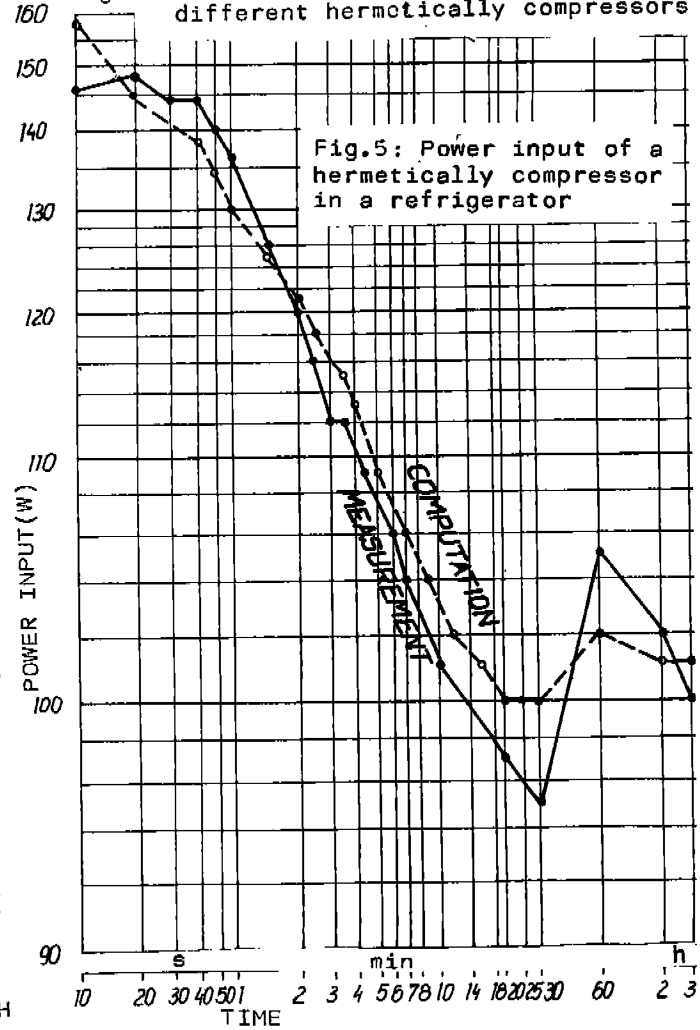

Fig.5: Power input of a hermetically compressor in a refrigerator

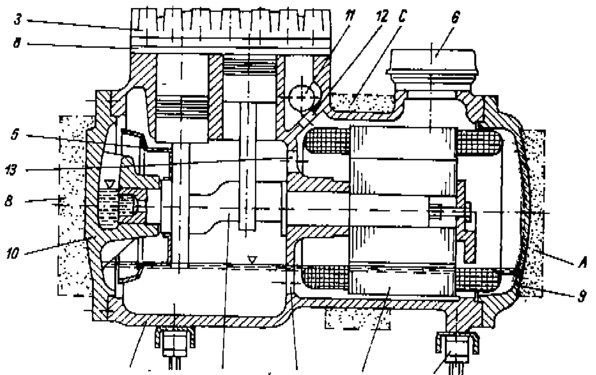

Fig.6:Semi-hermetically compressor 50-2
A,B,C water cooling, 5 centrif.disk f.oil

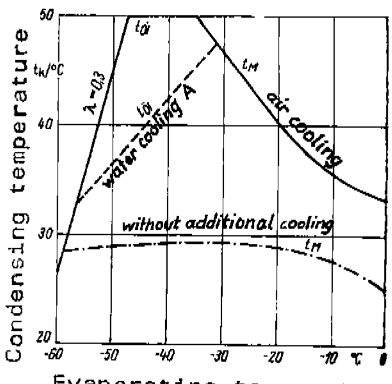

Fig.7: Limits of application with R22

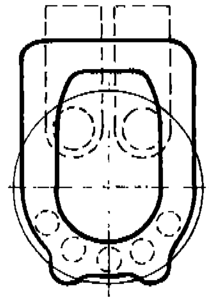

Fig.8: Reed suction and discharge valve

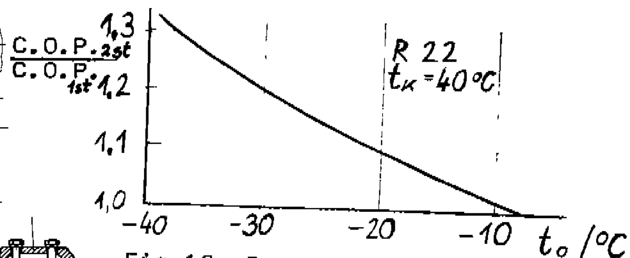

Fig.10: Improvement of C.O.P. in consequence of 2-stage compression

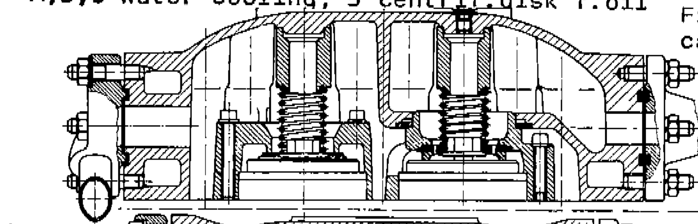
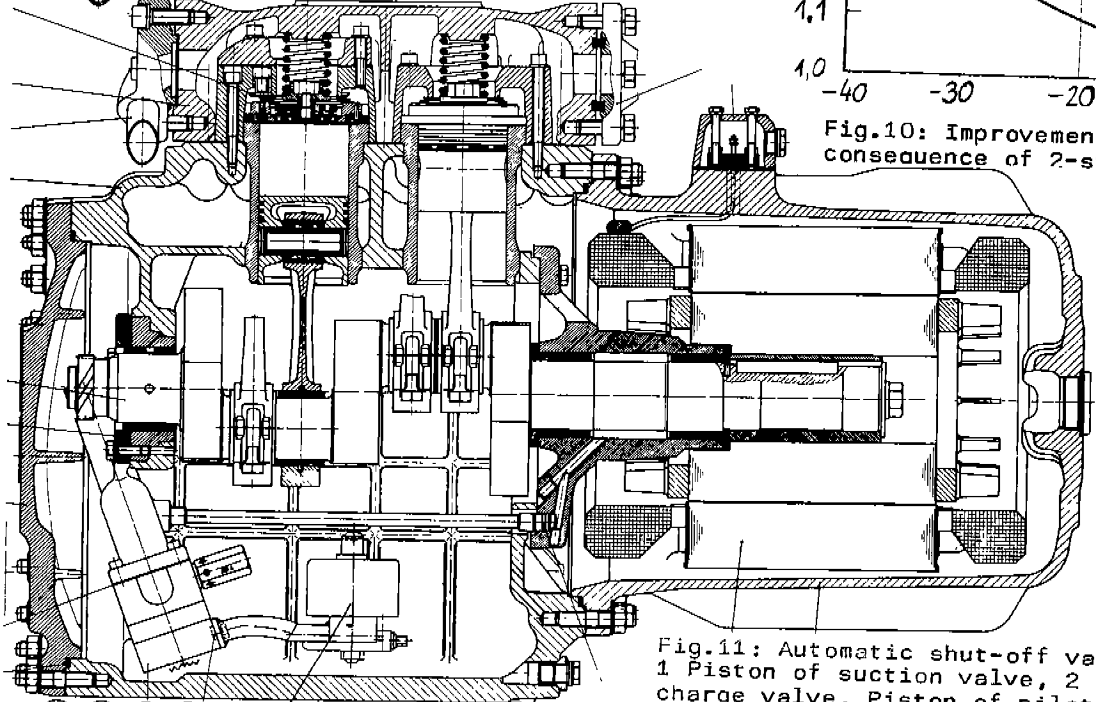

Fig.9: Compressor type 60-4/2

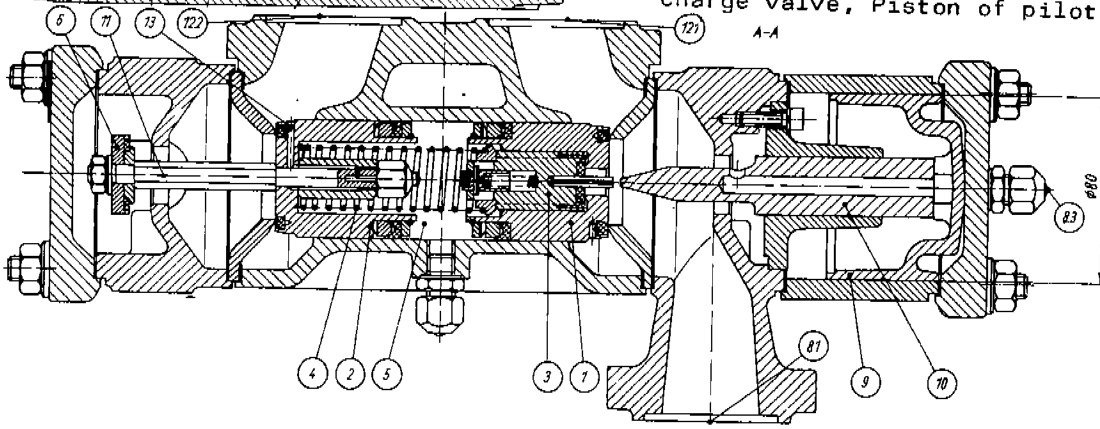

Fig.11: Automatic shut-off valve
1 Piston of suction valve, 2 Piston of discharge valve, Piston of pilot valve,6 By-pass valve,81 Suction line to compr.
82 Discharge line from compr.
83 Oil line
9 Hydr.operated piston
121 Suction line from evap.
122 Discharge line to cond.
13 Pressure equalizing

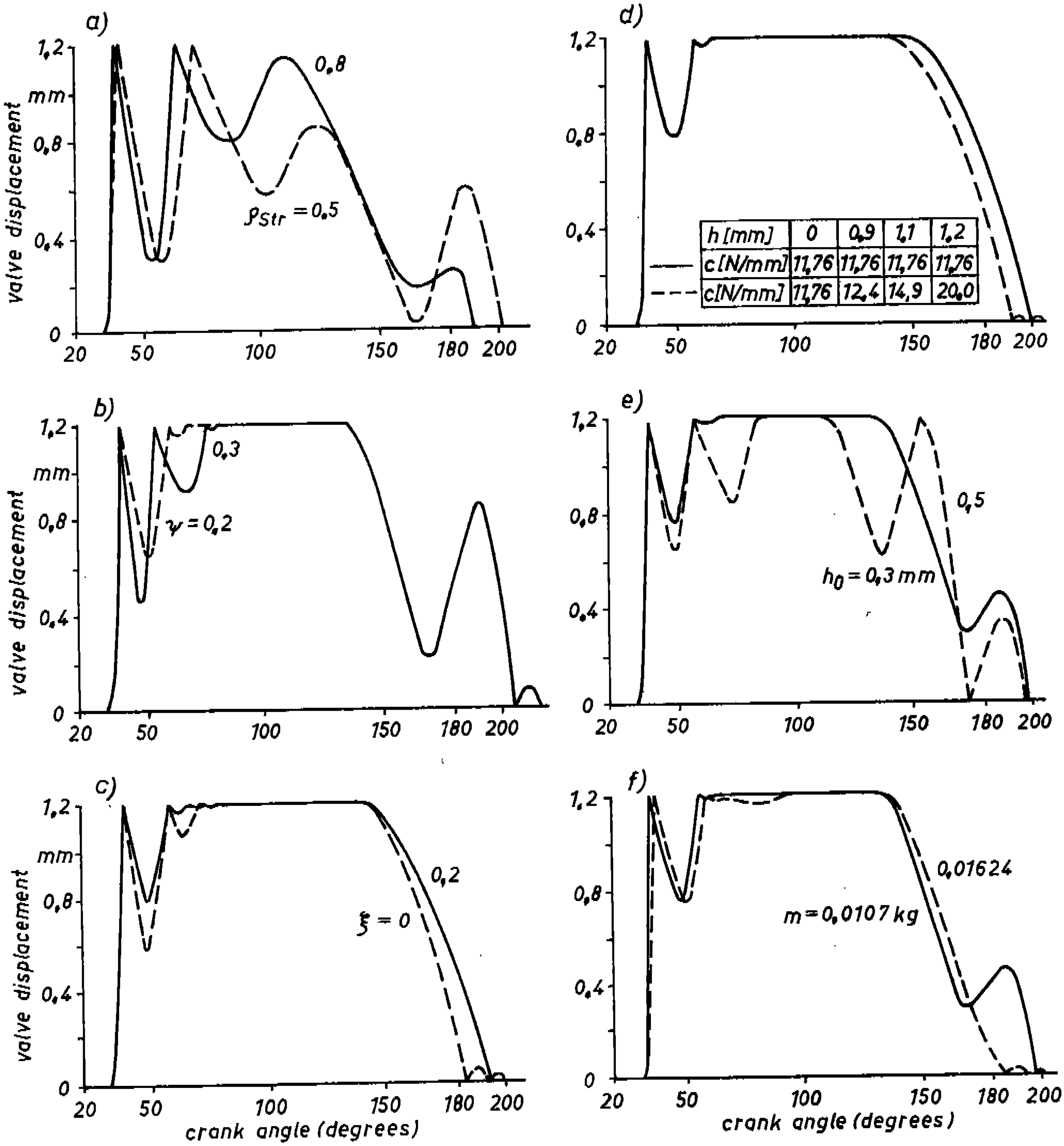

Fig. 12 Suction valve displacement
a) with different ϱ_{Str} b) with different ψ c) with different ξ
d) with different c e) with different h_0 f) with different m

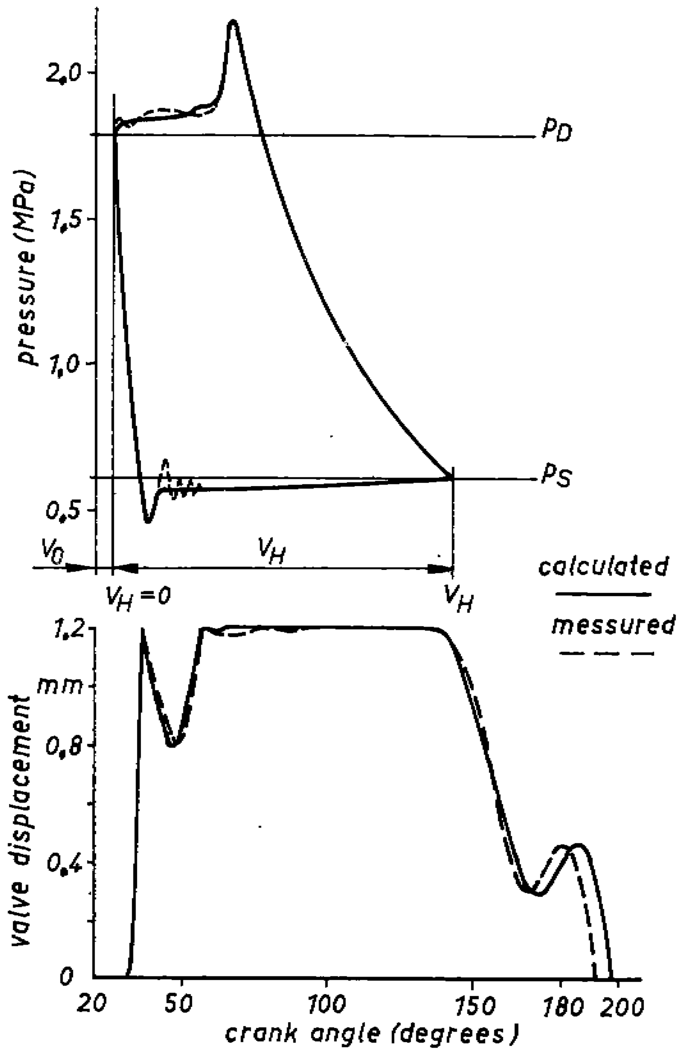

Fig.13 Comparison of messured and calculated
a) valve displacement diagrams
b) p, V - diagramms

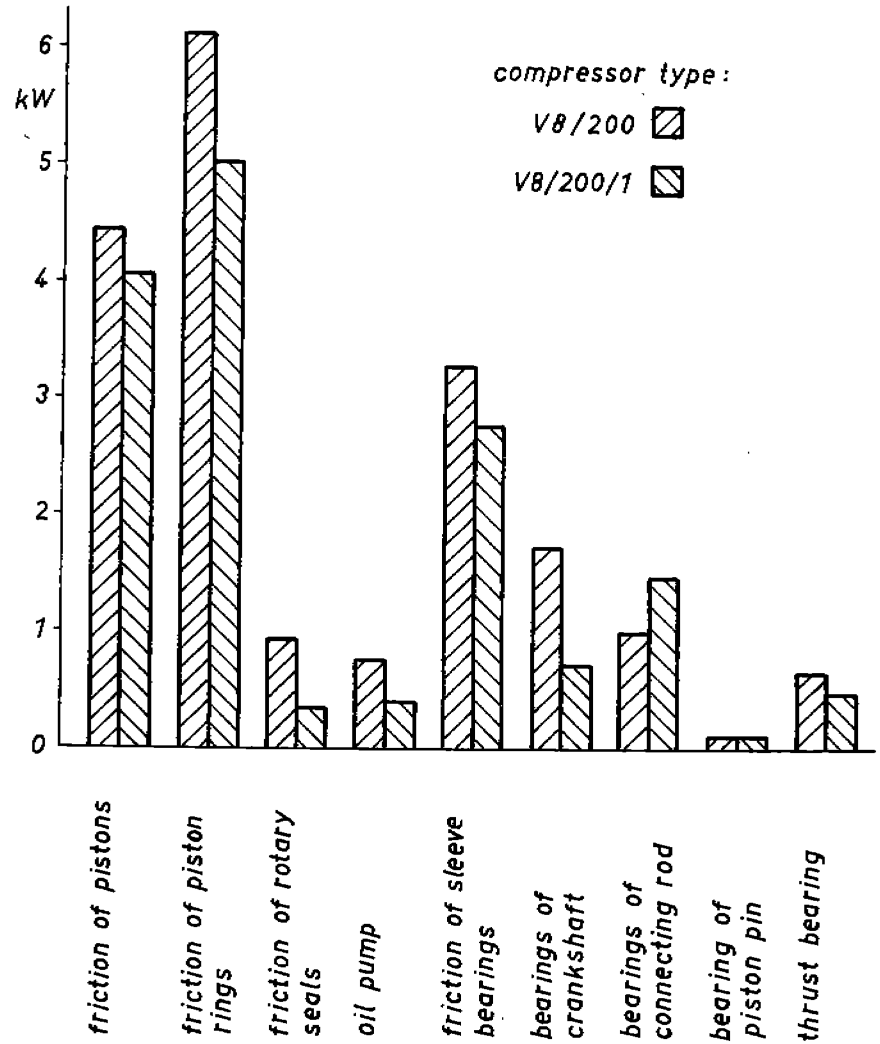

Fig. 14 Frictional energy losses
at conditions $t_0 = +10°C$, $t = +55°C$

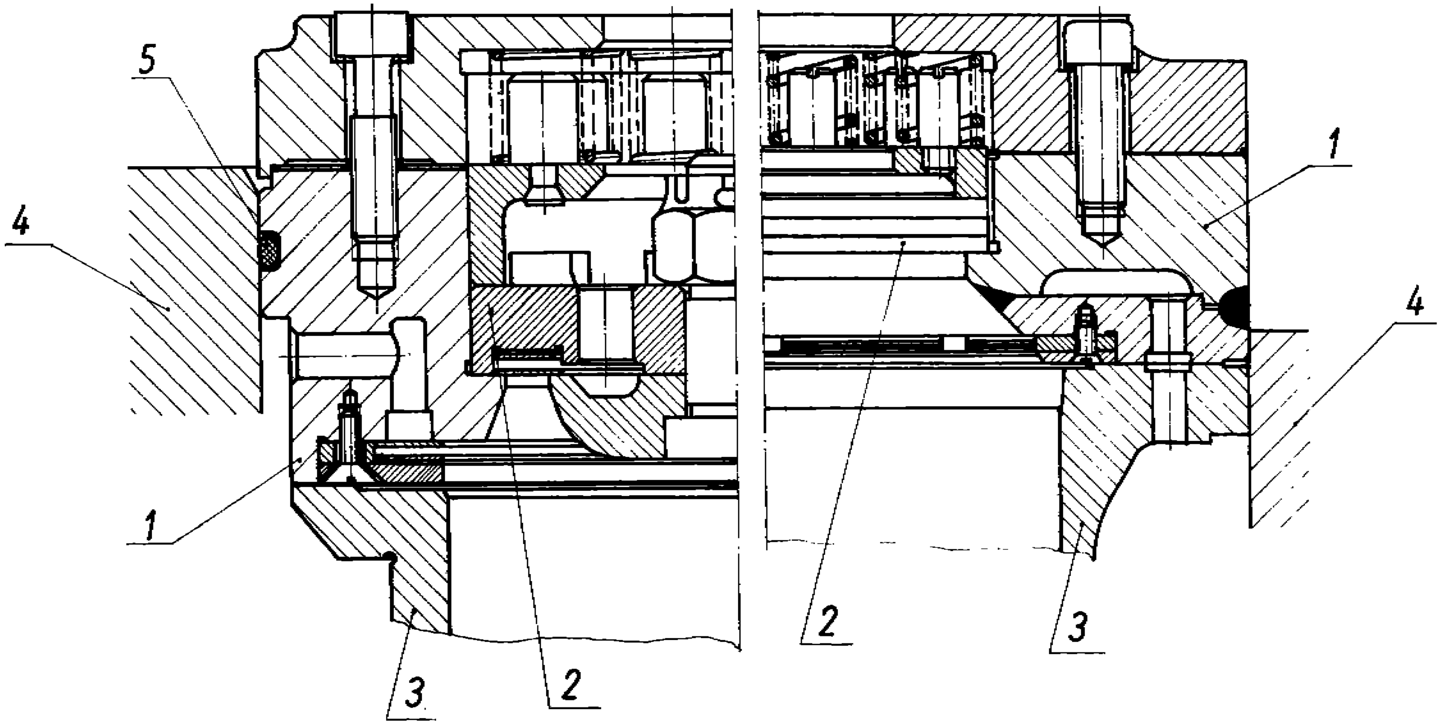

Fig. 15a Valve fixing, BR5 *Fig. 15b Valve fixing, BR2*

1-Suction valve 2-Discharge valve 3-Cylinder 4-Crank-case
5-Sealing rubber ring